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**SPACE SIMULATION FACILITIES
PROVIDING A STABLE THERMAL VACUUM FACILITY**

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ABSTRACT

CBI has recently constructed the Intermediate Thermal Vacuum Facility for Grumman Aerospace Corporation at Bethpage, N.Y. Built as a corporate facility, the installation will first be used on the Boost Surveillance and Tracking System (BSTS) program. It will also be used to develop and test other sensor systems. Completed in March 1989, the facility is composed of a 79.3 cubic meter (2800 ft³) stainless steel chamber, a liquid nitrogen cooled shroud and a pumping system to enable evacuation from one atmosphere to 2.7×10^{-5} Pascals (2×10^{-7} torr) in 24 hours or less. The horizontal chamber has a horseshoe shaped cross section and is supported on pneumatic isolators for vibration isolation. The chamber structure was required to have a minimum natural frequency of 50 Hz or higher. Through the use of consultants and in-house expertise, CBI designed the foundation configuration, pneumatic isolation system, and stiffened shell to meet the stability and stiffness requirements. The design process included measurement of the ambient ground vibrations, analysis of various foundation test article support configurations, design and analysis of the chamber shell and modal testing of the chamber shell. A detailed 3-dimensional finite element analysis was made in the design stage to predict the lowest three natural frequencies and mode shapes and to identify local vibrating components. The modal testing of the chamber shell was made on site with the isolated chamber in place including all piping and attachments. This paper describes the design process used by CBI and compares the results of the finite element analysis to the results of the field modal testing and analysis for the 3 lowest natural frequencies and mode shapes. It also presents concepts for stiffening large steel structures and methods to improve test article stability in large space simulation facilities.

INTRODUCTION

Space simulation facilities often require extremely stable test article support systems. It is increasingly common for owners to specify the stiffness of the chamber and/or maximum allowable displacements of the test article support system. Chamber stiffness is typically specified as a minimum dynamic natural frequency. CBI recently completed Grumman's BSTS thermal vacuum facility which had both a specified minimum chamber natural frequency and a maximum test article support displacement. To meet these stability requirements,

CBI developed and coordinated the entire design, analysis, procurement, installation, and testing of the facility.

Ambient ground vibrations were measured and applied to finite element models of 5 potential chamber and test article support configurations. The different configurations contained various pneumatic isolation systems for the support of the chamber and/or test platform. A three dimensional model of the selected configuration was analyzed to produce the most economical design of the entire support system, including the seismic mass foundation, while meeting stability requirements. In addition, the chamber shell was designed to have a minimum natural frequency of 50 Hertz.

The preliminary chamber design was based on classical solutions of geometric sections and panels. A detailed finite element model of the chamber was analyzed to determine the significant structural frequencies. The analysis predicted a minimum natural frequency of 53 Hertz. The analysis also identified local vibrating elements which were subsequently restrained with additional reinforcement. Once installed, the lowest three chamber natural frequencies were field measured through modal analysis. The measure results compared well with the finite element results. In addition, the measured test article displacements were within the minimum specified.

FACILITY DESCRIPTION

VACUUM CHAMBER

As shown in Figure 1, the chamber has a horseshoe shaped cross section in a vertical plane with a flat bottom. The shell is 6.9M (22'-6) long. The inside radius of the upper cylindrical section is 1.9M (6'-2). The vertical height from the floor to the top of the cylinder is 3.9M (11'-8). The width of the flat floor is 2.7M (9'-0). One end of the chamber is fixed and contains ports for three 889mm (35") cryopumps and a 1.1M x 1.8M (3'-6" x 6'-0") personnel door. The other end of the chamber is a full opening door. All interior surfaces are A240 Type 304L stainless steel with a #4 finish. The test article can be supported from ten hard points. The hard points are in five pairs which straddle the chamber centerline and are spaced 1.2m (4' - 0) apart. The chamber is designed per ASME Section VIII Division 1 where applicable. The chamber is supported on six pneumatic isolators. The six isolators are connected in three pairs to form a 3 point system. Each pair has a height sensor servo valve to control the isolator height. The isolators are supported on a 136,000 kg (300,000 lb) concrete seismic mass which is 4.6m (15') wide, 7.6M (25') long, and 1.8M (6') deep with cut outs on both sides for the isolators. The chamber foundation is independent from all other foundations to minimize the transmission of equipment vibrations to the chamber foundation.

PUMPING SYSTEM

The chamber can be evacuated from atmospheric pressure to 2.7×10^{-5} Pa (2×10^{-7} torr) in less than 24 hours. CBI installed two Leybold Heraeus WAU-2000/S400F blower/roughing trains and two nominal 35" diameter High Vacuum Equipment Corporation cryopumps with vibration isolated cold heads. The system also includes a Balzer 500 l/s turbomolecular pump for helium removal during high vacuum and leak checking. The cryopumps were installed nearly flush with the rear head plate to minimize conductance losses.

THERMAL SYSTEM

The facility contains a 100% optically dense aluminum shroud which is composed of six sections. The shroud provides a 3353 mm (11'-0") diameter x 6096mm (20'-0") long test envelope. The LN_2/GN_2 shroud can be maintained at a $-190^\circ\text{C} \pm 5^\circ\text{C}$ ($-310^\circ\text{F} \pm 10^\circ\text{F}$) temperature with a uniformly distributed heat load of 5 kW. In addition, the shroud can maintain any temperature between -171°C to 116°C (-275°F to $+240^\circ\text{F}$) within $\pm 3^\circ\text{C}$ ($\pm 5^\circ\text{F}$). The design pressure of the shroud is 1034 kPa (150 psig) with design temperatures of -195°C and 149°C (-320°F and $+300^\circ\text{F}$).

SYSTEM PSD VIBRATION ANALYSIS

SUPPORT SYSTEM REQUIREMENTS

During the proposal stage of the facility, CBI offered to assist Grumman in the design of the test article support system. CBI proposed that the entire system be studied from the soil to the test article platform. The goal of the study was to economically design a system which would limit the relative displacement of the test article platform to one micron or less. To accomplish this goal, CBI proposed and received acceptance for a Power Spectral Density, PSD, seismic study. The support system design included the design of the test article platform isolation and attachment details as well as the chamber support system. CBI provided a site survey specification which was used by Grumman to acquire a PSD seismic survey of the site. CBI hired the Ralph M. Parsons Company, Pasadena, California to assist CBI in the selection of a suitable support system configuration through a PSD study. The analysis was based on the PSD for the site, the chamber details, and the basic characteristics of the test platform.

PSD METHOD OF ANALYSIS

Initially, five sets of two dimensional analyses were conducted to determine which support configurations of the vessel and internal platform satisfied the relative motion requirements. Isolation system frequencies in the range of 0.5 Hz to 2.5Hz were considered in the two dimensional analysis. The five configurations examined are shown in Figures 7 through 11. These two dimensional analyses demonstrated that any of the five combinations of vacuum chamber

and/or internal platform isolation systems would satisfy the one micron relative motion limitation.

RELATIVE MOTIONS ON THE INTERNAL PLATFORM

Case	Vertical Displacement (micron)	Angular Rotation (nano-radians)
A	.216	135
B	.180	150
C	.204	147
D	.312	173
E	.048	47

The most simple and economical configuration studied consisted of a pneumatically isolated chamber with the test platform rigidly attached to the chamber. In addition to the economy of keeping isolators outside of the chamber, this configuration facilitates future modifications if additional stability were required. Thus, this configuration was chosen for further study.

A three dimensional finite element model was then developed which included the vacuum chamber, the external chamber isolation system, and the internal platform. The three dimensional model assumed that the chamber was isolated from the ground using six pneumatic isolators with a nominal isolation frequency of 1.0 Hz. The internal platform was assumed to be connected directly to the chamber structure without using an isolation system. The models of the chamber and platform were analyzed separately to ensure that their stiffnesses were accurately represented. This chamber mode gave a first model frequency of 60 Hz which compared well with the preliminary results. At the time of the analysis, the platform had not been designed but it's approximate mass and stiffness were known. The possible platforms first three modal frequencies were 42, 61, and 102 Hz which was consistent with Grumman's estimated first mode frequency for the platform. When the chamber and platform were confined and supported on the 1 Hz isolation system, the first two modal frequencies were 37 Hz and 44.6 Hz. The analysis identified 33 modal frequencies of the chamber platform system below 150 Hz. These 33 modes were used to determine the overall modal response.

The stiffnesses of each of the mechanical connections to the chamber penetrations were included in the three dimensional finite element model. The mechanical connections consist of flex hose and pipe bellows. The ground motion vibrations were applied to the chamber through the connections to the vessel penetrations as well as through the six external chamber isolators. The stiffness of the nozzle connections are shown in the table below. N-S is the north-south direction which is along the horizontal longitudinal axis of the chamber. E-W is the east-west direction.

NOZZLE STIFFNESS

NOZZLE #	QUANTITY No.	N-S STIFFNESS Kg/mm	E-W STIFFNESS Kg/mm	VERT STIFFNESS Kg/mm
1	2	1.34	1.79	1.34
2	1	5.84	5.57	5.57
6	2	1.07	1.07	1.07
7	5	2.68	2.68	2.68
9	1	2.55	2.55	3.39
10	1	7.04	5.18	5.18
11	1	2.55	2.55	3.39
14	1	0.89	0.89	0.89

The input PSD spectra was 5.0×10^{-10} g^2/Hz over the frequency range of .1 Hz to 10 Hz and 5.0×10^{-9} g^2/Hz over the frequency range of 10 Hz to 300 Hz. The PSD spectra were applied in both the vertical and horizontal directions simultaneously.

The PSD analysis was performed on the Stardyne computer program by the Boeing Computer Service in Seattle, Washington. The system model contained 267 nodes, 291 beam elements, 34 triangular plate elements, and 175 quadrilateral plate elements.

PSD ANALYSES RESULTS

The two dimensional model of the selected configuration produced a minimum displacement of .05 microns while the three dimensional analysis maximum displacement was .94 microns. The three dimensional analysis included the stiffness and vibration input of the major chamber attachments which may account for the increase in the relative displacements for the 3-D model.

PSD SUMMARY

The results of the PSD study indicated that the use of external isolators would meet the stability requirements. The high natural frequencies of the chamber and platform system increases the performance of the isolation system. The primary method of vibration attenuation of the isolation system is to have the natural frequency of the isolation system as low as possible and to have the natural frequency of the isolated structure as high as possible. Generally, structural frequencies should exceed the isolation frequency by a factor for 10 to 15. In this case, this ratio was approximately 37 which results in a very large attenuation of the ground vibrations.

FOUNDATIONS AND ISOLATION SYSTEM DESIGN

A large concrete seismic mass was used to support the chamber isolation system. The 160,000 kg (300,000 lb) seismic mass is roughly 5 times the weight of the entire chamber. The design is based on the concept that the foundation should be rigid and

independent of surrounding foundations. A rigid foundation will tend to force the ground motions introduced into the base of each isolator to be in phase. The independent foundation will help prevent equipment vibrations from direct transmission to the chamber foundation. The concrete foundation and surrounding sandy soil provide excellent damping to improve the vibration attenuation.

The isolation system was procured and installed by CBI based on the results of the PSD study. The isolators are 635 mm (25") in diameter and 1194 mm (47") high. Each isolator has a 9072 kg (20,000 pound) capacity and lifts the chamber approximately 6 mm when activated. The isolators were connected in three pairs to form a three point support system. The height sensor servo valves are capable of maintaining each of the three isolator pairs within .13mm of each other to provide an active vibration isolation system. The isolators operate with nitrogen gas between 552 and 690 kPa (80 and 100 psig).

SHELL STRUCTURE DESIGN PROCEDURE

The chamber shape was selected by CBI to provide the flexibility required for the support configuration, proper frequency response, transportability, and economy. In the preliminary stage, equivalent cylinders and flat panels were analyzed with classical solutions to determine the theoretical natural frequencies. A similarly stiffened cylinder was analyzed by a CBI Technical Services program entitled "General Shell of Revolution Stress Analysis - Dynamic Version". This is a multi-segment numerical integration procedure developed by A. Kalnins and presented in the Journal of Applied Mechanics, September 1964. A 6.9M long cylinder with simply supported ends and identical circumferential stiffening was chosen which had a minimum natural frequency of 60 Hertz. The flat portions of the chamber, namely the lower shell and heads, were compared to the classical solutions for flat plates as presented by Blevins in his text entitled "Natural Frequencies and Mode Shapes". These design elements were then combined to provide the preliminary design. The design was checked per ASME Section VIII, Division 1. Procurement and fabrication of the chamber shell was initiated based on this design.

CHAMBER FREQUENCY ANALYSIS

METHOD OF ANALYSIS

The preliminary design was then modeled to determine the lowest dynamic frequencies and mode shapes. This analysis was done entirely in-house by CBI analysts using programs either developed by CBI or commercially available. The approach was to use the finite element method, FEM, to analyze the chamber. The lowest natural frequencies and mode shapes were calculated using a CBI Technical Services computer program, entitled "Automatic Dynamic Incremental Nonlinear Analysis (ADINA)." The ADINA program is a general purpose computer code for the linear and nonlinear, static and dynamic, displacement and stress analysis of solids, structures and fluid-structure

systems. The ADINA program is licensed from ADINA R&D, Inc. of Watertown, Massachusetts and is verified for use on CBI's IBM 4381 computer through the solution of over 75 linear and over 70 nonlinear verification problems.

GEOMETRY & MODELING

The chamber is geometrically symmetric about a vertical plane which runs along the length and passes through the center of the chamber. The structure is also symmetric about a vertical plane through the cross section at the mid length except for the ends of the chamber. The chamber has a full cross section stiffened closure door at one end and a stiffened fixed head at the other end. The stiffening patterns on the door and head are completely different. In addition, the fixed head contains large openings for the cryopumps and a personnel access door. To enable the use of two planes for symmetry, two chamber configurations were analyzed with 1/4 models. The two models represent chambers with identical ends, either two full doors or two fixed heads. Although two models had to be analyzed, the final results were finite element models with good detail and limited size due to two planes of symmetry.

The element types used to model the chamber were sixteen-noded shell elements, four-noded shell elements, four-noded isoparametric beam elements and two-noded beam elements. All stiffening was discretely modelled. Rigid links, constraint equations or contiguous modelling methods were used to connect the stiffening to the pressure boundary. The "vessel-door" model contained about 1350 nodes, 460 elements, and 5800 degrees of freedom. The "vessel-head" model contained about 1430 nodes, 710 elements, and 6500 degrees of freedom. Figures 2 and 3 show finite element plots of the "vessel-door" and "vessel-head" models, respectively.

Constraint equations were used in the "vessel-door" model to connect the chamber door to the vessel shell. Only displacement compatibility was enforced across the door to shell interface. This permitted independent rotation of the door and shell interface. This accurately represented the pressure seated closure. In the "vessel-head" model, the same modelling approach was used to connect the personnel access door to the fixed head.

The high vacuum valves and cryopumps were assumed rigid. The vessel contains penetrations for three cryopumps. Cryopumps are installed on the two outer ports only. The center port is blanked off and is for future use. The chamber was modeled with the third cryopump in place. Their mass was connected to the fixed head using rigid links. Items such as closure door hinges and mounting brackets, small penetrations and viewports, and access holes cut through the webs of stiffeners were assumed insignificant and were not modelled. The mass of the internal shroud was assigned to the model nodes nearest the shroud support points.

The chamber isolation supports were included in the model. Each

isolator was modelled using a set of four beams. The beams were oriented vertically. They were fixed at the bottom and pinned at the top where they connect to the chamber bottom. The assumed stiffness values used for each isolator were 1500#/in vertical and 750#/in lateral.

MATERIAL PROPERTIES

Linear elastic material behavior was modelled. The important material properties are modulus of elasticity, Poisson's ratio, and weight density. The vacuum chamber is made of two materials. The pressure boundary material in contact with the vacuum is A240 TP 304L stainless steel. The stiffening and non-pressure parts are A36 carbon steel.

LOADING AND BOUNDARY CONDITIONS

Only inertial loading was considered. The mass of the chamber shell, door, head, personnel access door, and stiffening was included by discretely modelling these components. The mass of the high vacuum valves, cryopumps and internal shroud was included through the use of concentrated nodal mass input.

Each model was run using four different sets of boundary conditions along the planes of geometric symmetry. These were: symmetry-symmetry, symmetry-antisymmetry, antisymmetry-symmetry, and antisymmetry-antisymmetry.

CHAMBER FEM RESULTS

The first mode has a frequency of 53 Hertz and is shown in Figure 4. The mode shape is essentially an ovaling of the central portion of the chamber shell about the vertical & horizontal center lines. The mode shape is symmetric about the vertical planes of symmetry along the length for the chamber and at the mid-length of the chamber. It is shown using the mesh from the vessel door model.

The second mode has a frequency of 63 Hertz and is shown in Figure 5. The mode shape is essentially an ovaling of the central portion of the chamber shell about a set of axes inclined approximately 45°. The mode shape is anti-symmetric about the vertical plane of symmetry which runs along the length of the vessel and is symmetric about the vertical plane of symmetry at the mid-length of the chamber. It is shown using the mesh from the vessel door model.

Three modes are closely spaced in the frequency range of 68 to 72 Hertz. The only mode of these three which does not include movement of the center cryopump has a frequency of 72 Hertz and is shown in Figure 6. Its mode shape is movement of the cylindrical portion of the shell between the end and middle circumferential stiffeners. This mode shape is symmetrical about the vertical plane of symmetry which runs the length of the chamber and is anti-symmetric about the vertical plane of symmetry at the chamber mid-length across its

width.

The FEM results were consistent with the theoretical dynamic results of the preliminary design. As stated earlier, an equivalent cylinder was analyzed and found to have an $n=2$ mode frequency of approximately 60 Hz. The $n=2$ mode is an ovaling of the cross section which will take place at any angle around a cylinder. By using an irregular cross section like a horseshoe, two $n=2$ modes are developed which occur at consistent orientations. Both the FEM analysis and the modal analysis identified two $n=2$ modes with frequencies of approximately 50 Hz and 63 Hz. The test platform supports are located near the node lines for the lower $n=2$ mode and all support points move in phase which minimizes the effect of the lowest chamber mode on the test article stability.

The initial configuration of the chamber was slightly modified due to the results of the FEM analysis. The FEM analysis identified local stiffener modes whose natural frequencies were below fifty Hertz. The modes were local lateral displacement of the stiffener flanges. These modes were eliminated by adding gusset plates to prevent lateral movement of the stiffeners. In addition, the middle circumferential stiffener initially was composed of the 305 mm x 19 mm (12" x 3/4") bar only. The lowest shell mode with this stiffener was 51.5 Hertz. The addition of the 250mm x 31.7 mm (10" x 1 1/4") flange to this stiffener raised the first shell mode frequency to 53.5 Hertz. The chamber was built with this flange added to the middle circumferential stiffener.

MODAL ANALYSIS

FIELD VERIFICATION

The chamber, foundation, and isolation system were furnished and installed by CBI based on the results of the design and analysis. Once the chamber was completed including all attachments and systems, the chambers three lowest structural natural frequencies were determined by experimental modal analysis. Experimental modal analysis is the process of combining field measured dynamic response data to determine the dynamic behavior of a structure. It was used in this case to verify the FEM results. The modal analysis was performed at the direction of CBI by Wiss, Janney, Elstner Associates, Inc. of Northbrook, Illinois.

MODAL DATA ACQUISITION

Due to the longitudinal symmetry of the chamber, a grid pattern of 50 points was laid out on half of the chamber. Ten additional data acquisition points were established on the chamber door and fixed end. Dynamic frequency response function (FRF) data was obtained at all 60 points on the chamber with piezoelectric accelerometers. A hand-held impulse force hammer was used to strike the middle circumferential stiffener. The FRF data is basically a ratio of the output response to force input. Piezoelectric accelerometers, type

4335, manufactured by Bruel & Kjaer (B&K) were used for measuring acceleration response vibration amplitudes. The electrical signal from each accelerometer was routed to a B&K type 2635 charge amplifier. This amplifier was used to electrically condition the signal from the high impedance accelerometer. The hand-held impulse force hammer, type 086B05, manufactured by PCB Piezotronics, Inc., was used to apply a transient, dynamic force to the vacuum chamber. This impulse hammer produces an electrical signal proportional to the force output. It was routed to a PCB Piezotronics type 480D06 battery power unit for electrical conditioning.

The conditioned accelerometer and impulse force signals were analyzed and recorded on a Scientific Atlantic Model SD380Z, four channel spectrum analyzer. One channel received impulse force time history data from the modal hammer and two other channels received acceleration response data from two perpendicular accelerometers. Computer analyses were subsequently performed at the WJE laboratories using Star Modal Software (SMS) which is a product of Structural Measurement Systems Inc. This software uses frequency response function data to identify the modal properties for a structure. The modal parameter outputs of this software are frequency, damping, and mode shape characteristics. All frequency data are given in units for Hertz (Hz). Damping values are given in units of percentage relative to critical damping.

MODAL ANALYSIS RESULTS

The sum of the FRF amplitudes for the 50 shell positions are shown in the Figure 12. This preliminary summation identified three significant resonant frequency ranges of approximately 46-50 Hz, 62-66Hz, and 71-74 Hz. The peak at 60 Hz was determined to be electrical leakage noise. The Star Modal Software uses these selected frequency ranges and a curve fit process to identify the modal parameters. An analytical expression for a FRF is matched to the measurement data and modal parameters identified. A global curve fitting process developed by Structural Measurement Systems was used to determine their frequency and damping of the structure with the following results.

Mode Number	Frequency Hertz	Damping % Critical
1	48.0	.60
2	64.3	.44
3	72.3	.56

The mode shapes identified by the modal analysis confirmed or agreed with the results for the finite element analysis. The modal analysis confirmed the ovaling of the shell between the ends for modes 1 and 2. The modal data was insufficient to determine the third mode shape but agreed with the FEM results in that the only significant distortions were located in the cylindrical portion of the shell between the stiffeners.

The central stiffeners on the front door were found to have a resonant frequency of 54 Hz. The door plate frequency between the stiffeners was 120 Hz. Similarly, the lowest frequencies of the fixed end of the chamber were 120 and 136 Hz.

COMPARISON OF FEM ANALYSIS AND EXPERIMENTAL MODAL ANALYSIS

The table shown below compares the results of the FEM analysis and the Modal Analysis. Although the modal analysis is dependent on the curve fitting procedure used and the frequency ranges selected, the accuracy is within 2%.

MODE	FEM Hz	MODAL Hz
1	53	48.0
2	63	64.3
3	72	72.3

The frequency results for the first mode are within 10% of each other. The second and third mode values are within 2% and .5% of each other, respectively. In addition to the inherent approximations of the FEM, the following items may account for some of the discrepancy in the results.

Although there are a number of areas where the model differs from the actual structure, the greatest difference is the attachment of the full door to the chamber. The FEM analysis assumed that the door was seated against the chamber with a force sufficient to prevent displacement of the door relative to the vessel end. When under full vacuum, the door is seated against the chamber with a force in excess of 115,000 Kg. The modal analysis was conducted with the chamber at atmospheric pressure. C-clamps were used to hold the door against the chamber. Although firmly held in place, the clamping force was not sufficient to compress the o-ring seal sufficiently to cause metal to metal contact between the door and chamber. Thus, the door to shell contact was through the elastomeric o-ring. The mode shape identified by modal analysis did not indicate excessive displacement of the chamber at the door. However, the modal analysis mode shapes were not sufficiently detailed to identify anything more than the general mode shape. It should be noted that an equivalently stiffened cylinder with one open end has an ovaling first mode frequency of approximately 30 Hz.

The mass of the FEM model did not completely correspond to that of the chamber configuration when field tested. The locations of some attachment were not known when the chamber was analyzed and thus were not included. The chamber contains 14 nominal 300 mm diameter view ports. Ten of these are located 90° from the top of the chamber with five viewports on each side. These 50 Kg viewports are located at the location of maximum radial displacement for the first mode and at a node line for the second mode. The chamber also contains a platform at the top of the chamber which is at a location of maximum radial displacement for the first mode. The mass of the thermal

shroud was distributed over the entire chamber in the FEM analysis. The shroud was not in place during the modal analysis.

The FEM model discretely models the chamber stiffening but does contain some variations. The actual stiffening is stitch welded to the chamber but continuously attached in the model. These variations are minor and most likely not significant. It should be noted however that the third mode shape does not include stiffener displacement and was accurately predicted by the FEM analysis.

CONCLUSIONS

Through the use of design and analysis procedures, CBI successfully provided a thermal vacuum facility which exceeded the customer's stability requirements. The following points should be considered for future installations.

The specific ambient ground vibrations and soil conditions of a proposed facility must be determined and analyzed to accurately predict the facility stability. A Power Spectral Density analysis can be used to select and design the most economical support configuration. So important is the PSD analysis that CBI has developed the in-house capability to perform the PSD analysis since the completion of the Grumman facility.

Vibrations transmitted to the test article foundation must be limited. In this facility the test article was supported indirectly by the seismic mass. The seismic mass was independent of all other foundations. In addition, all the vibrating components associated with the system were supported on vibration isolated equipment skids even though supported on independent foundations.

Vibration inputs to the facility through penetrations and attachments can significantly increase the accelerations and resulting displacement of the test platform. Flexible hose and bellows were used on all attachments. Their stiffness was known and included in the analysis of the facility.

Detailed finite element models can successfully predict the mode shapes and corresponding frequencies of the chamber. Models must represent the final configuration as closely as possible. Basic geometric shapes can be used to identify general behavior of the facility. Naturally, the accuracy will be dependent on how closely the facility corresponds to the configurations studied.

Experimental modal analysis can be used to accurately determine the dynamic characteristics of a completed facility. Natural frequencies can be determined with a relatively small number of data points. Mode shapes are more difficult to determine and thus require closely spaced data points.

Knowledge of the vessel behavior can be used to limit both the test platform support vibrations and the vibration inputs through chamber

attachments. Supports and attachments should be placed at low frequency node lines. An irregular cross section or irregular stiffness is required to orient the modes and establish a consistent phase angle. Minimizing the effect of low frequency modes can be extremely beneficial for very large chambers which require tremendous stiffening to provide similarly high natural frequencies.

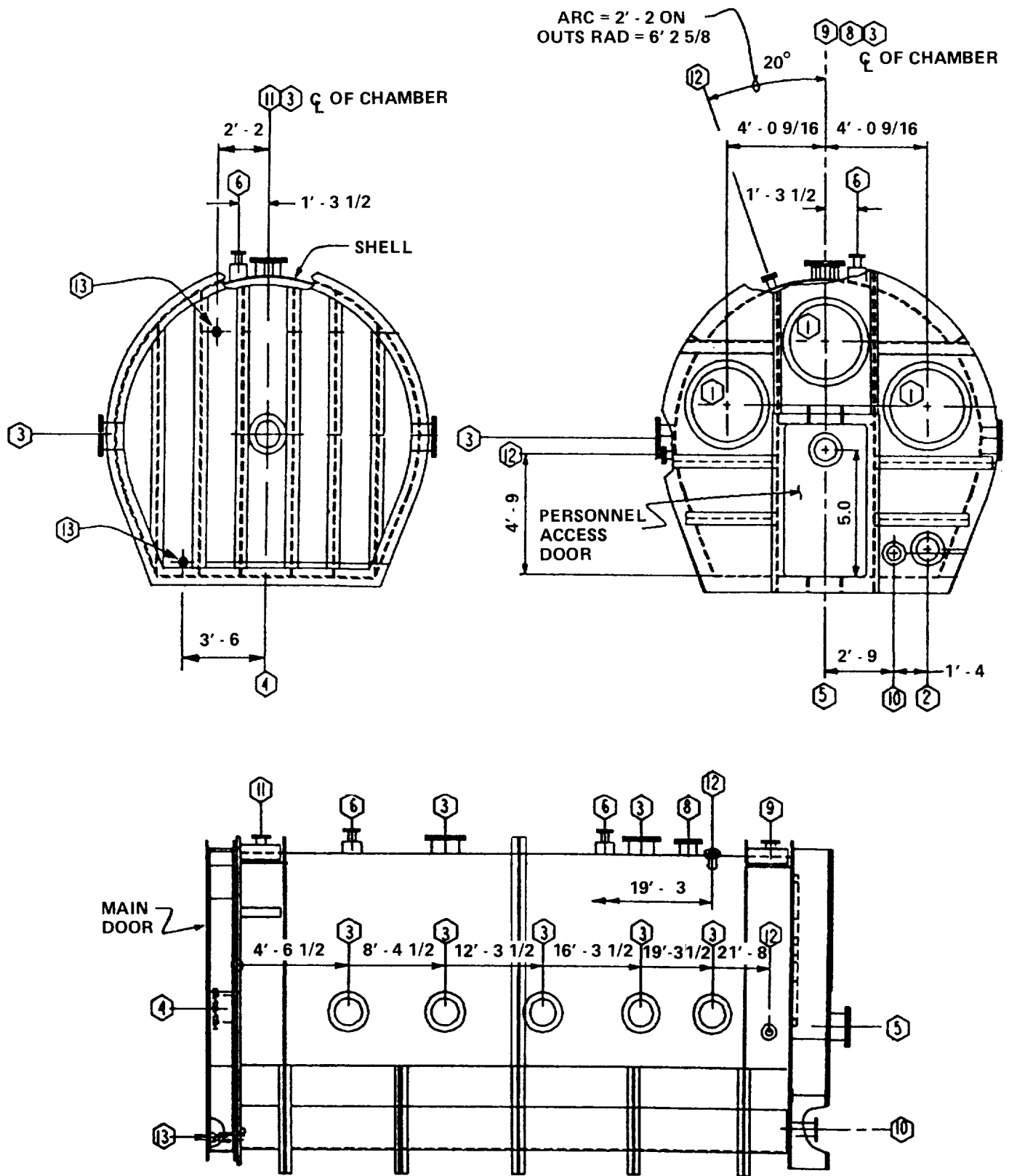


Figure 1. Chamber Configuration

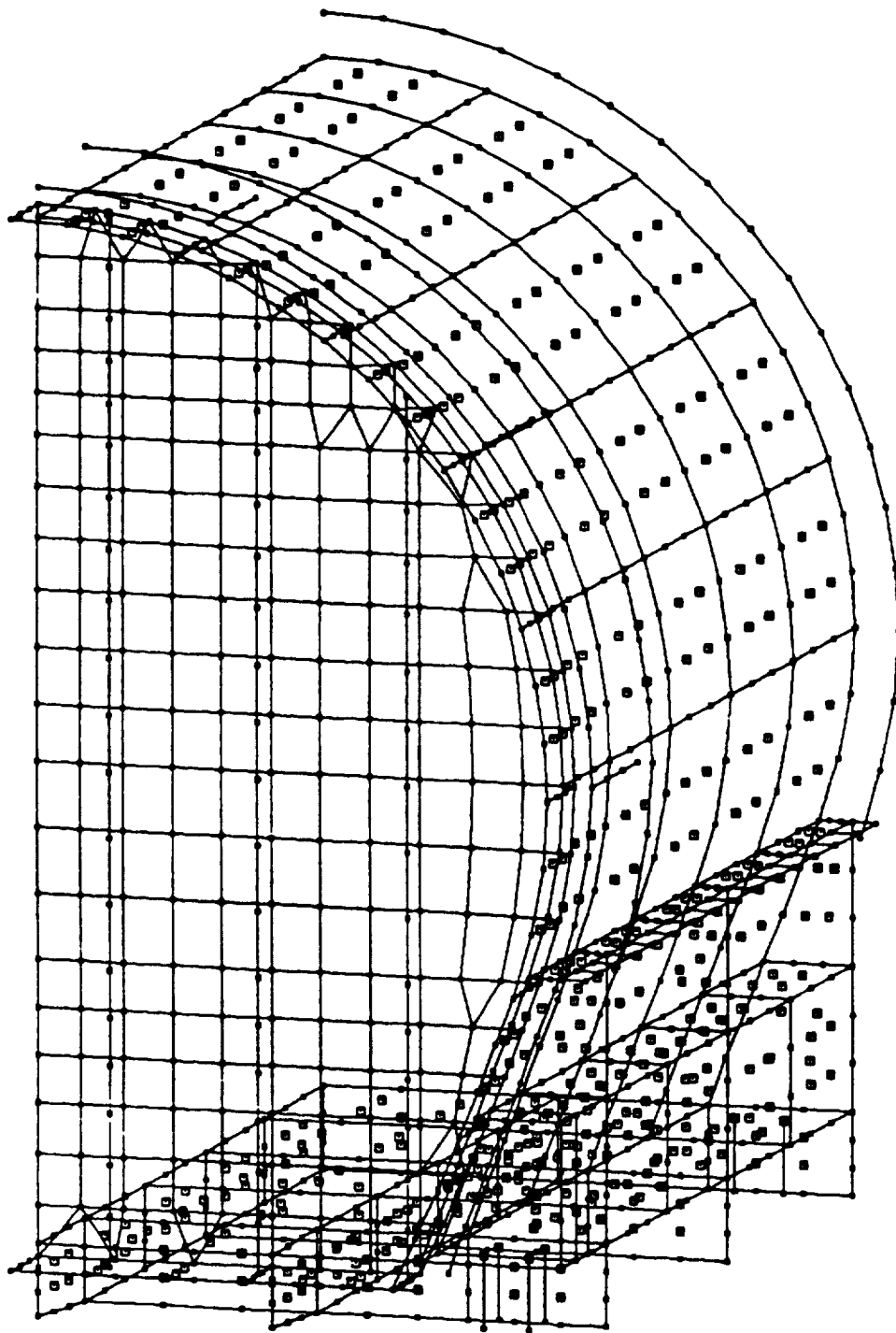


Figure 2. Vacuum Chamber Shell with Door — Quarter Model

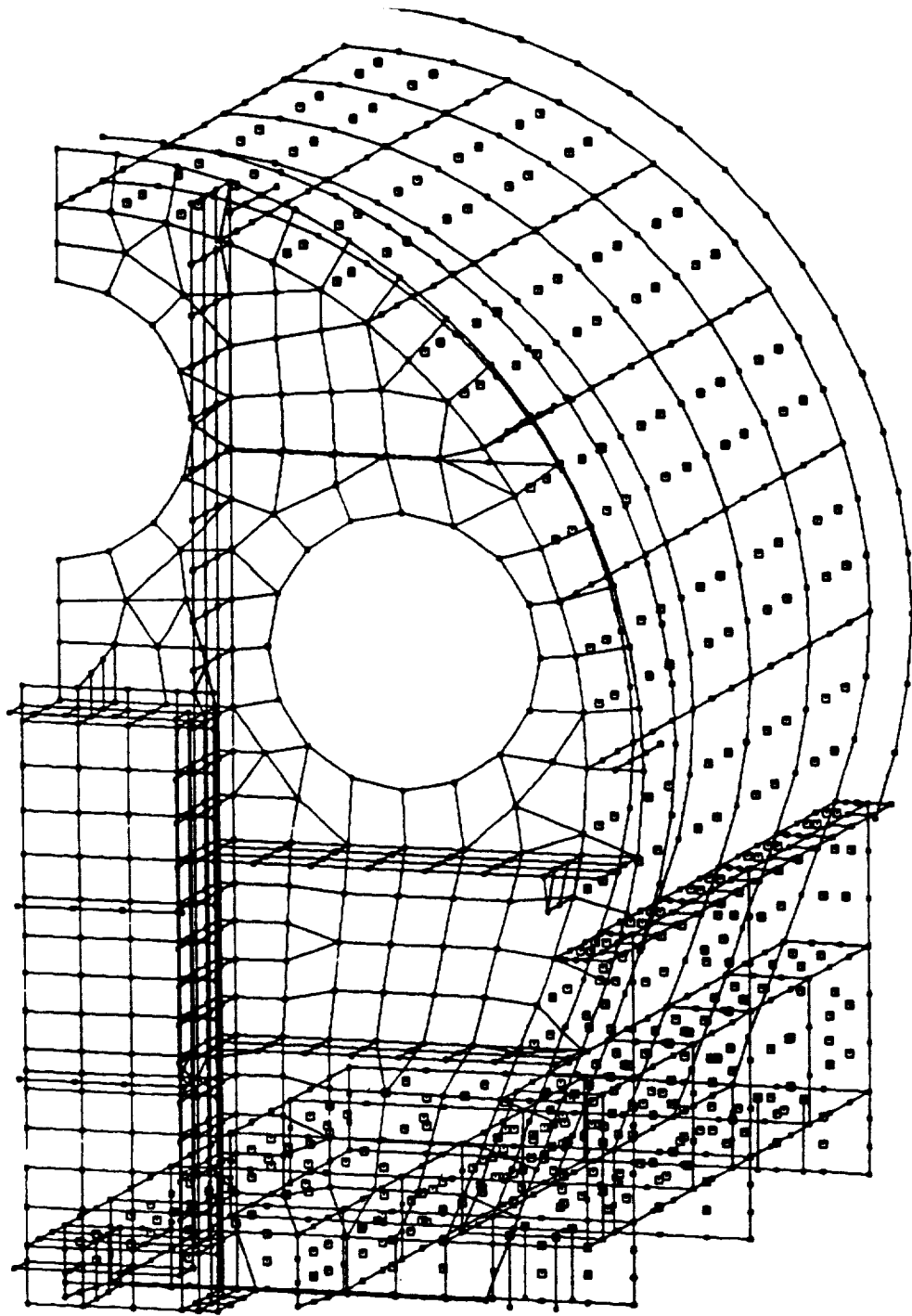


Figure 3. Vacuum Chamber Shell with Fixed Head – Quarter Model

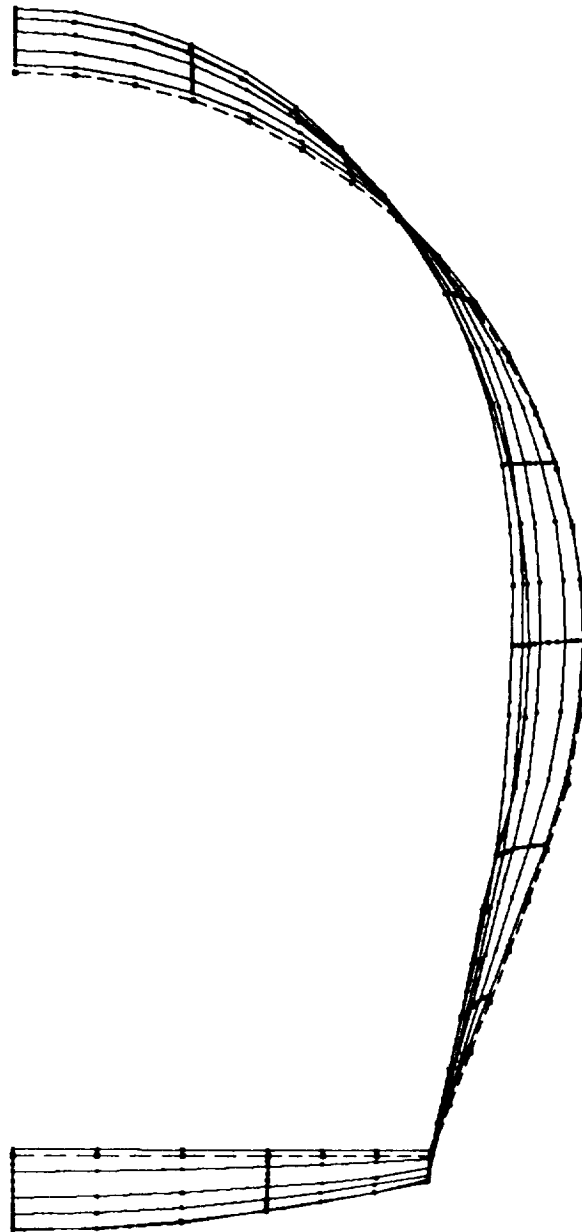


Figure 4. First Mode Frequency of the 1/4 Model – 53 Hz

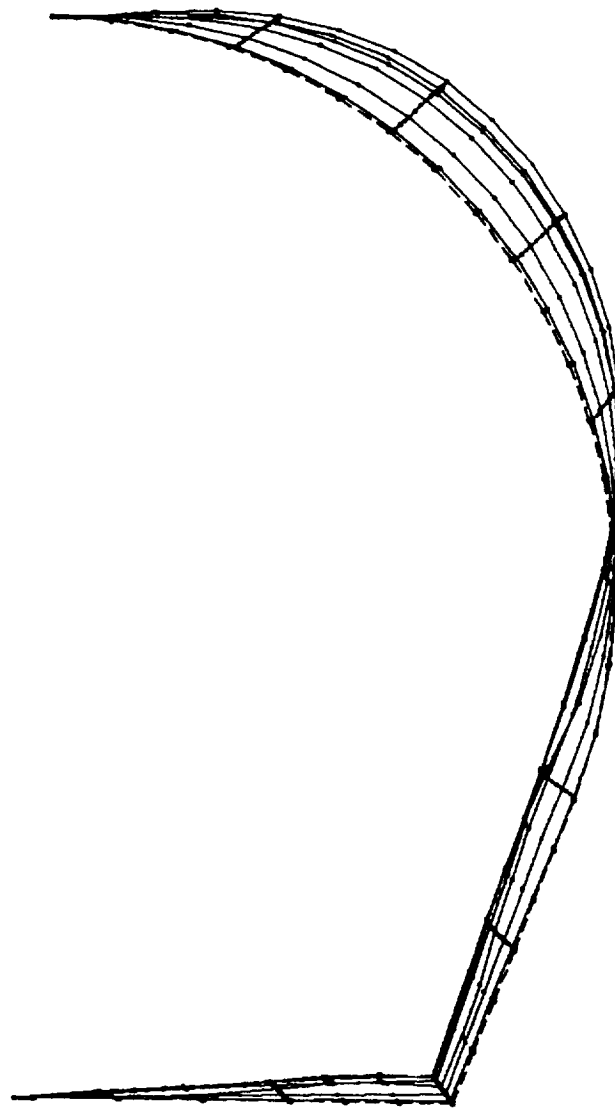


Figure 5. Second Mode Frequency of the 1/4 Model – 63 Hz

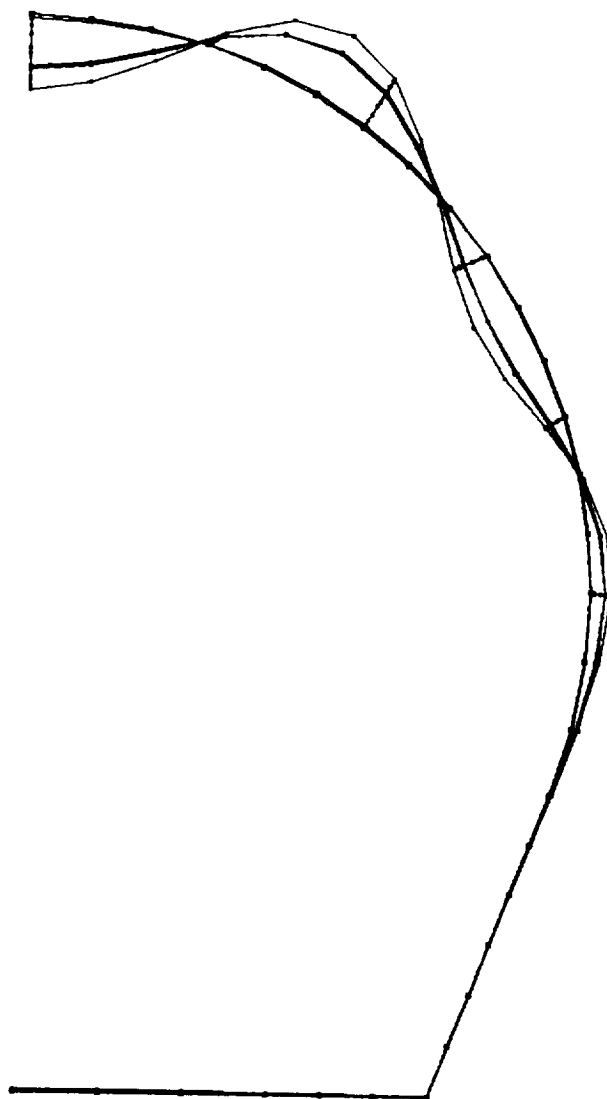


Figure 6. Third Mode Frequency of the 1/4 Model -- 72 Hz

- CASE A
- VACUUM CHAMBER SUPPORTED ON CONCRETE FOUNDATION
 - INTERNAL OPTICAL BENCH ISOLATION SYSTEM FREQUENCY = 1.0 Hz

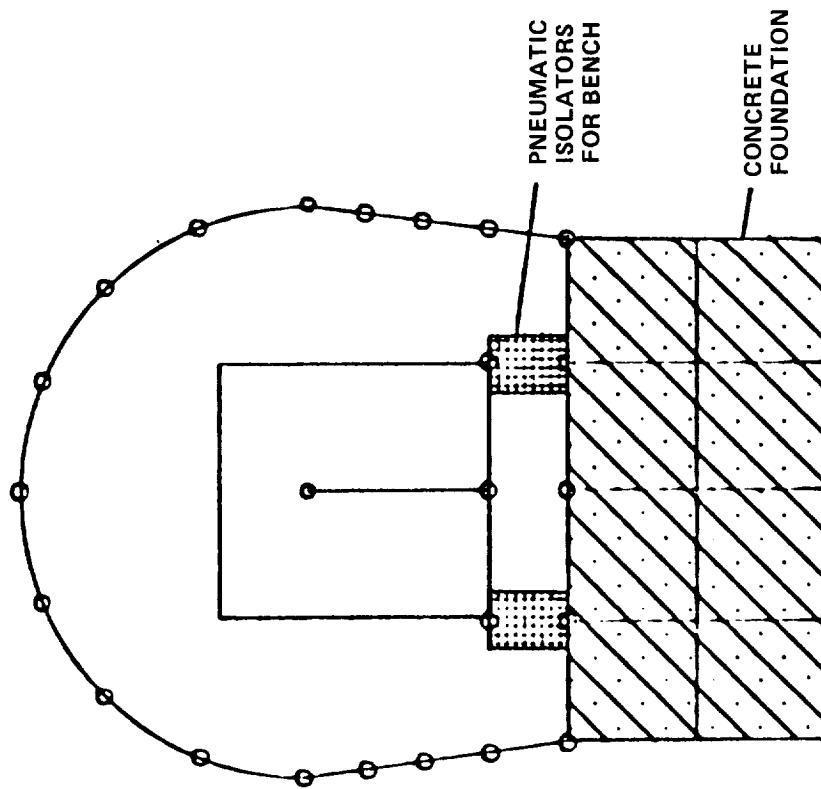


Figure 7. PSD 2-D Case A

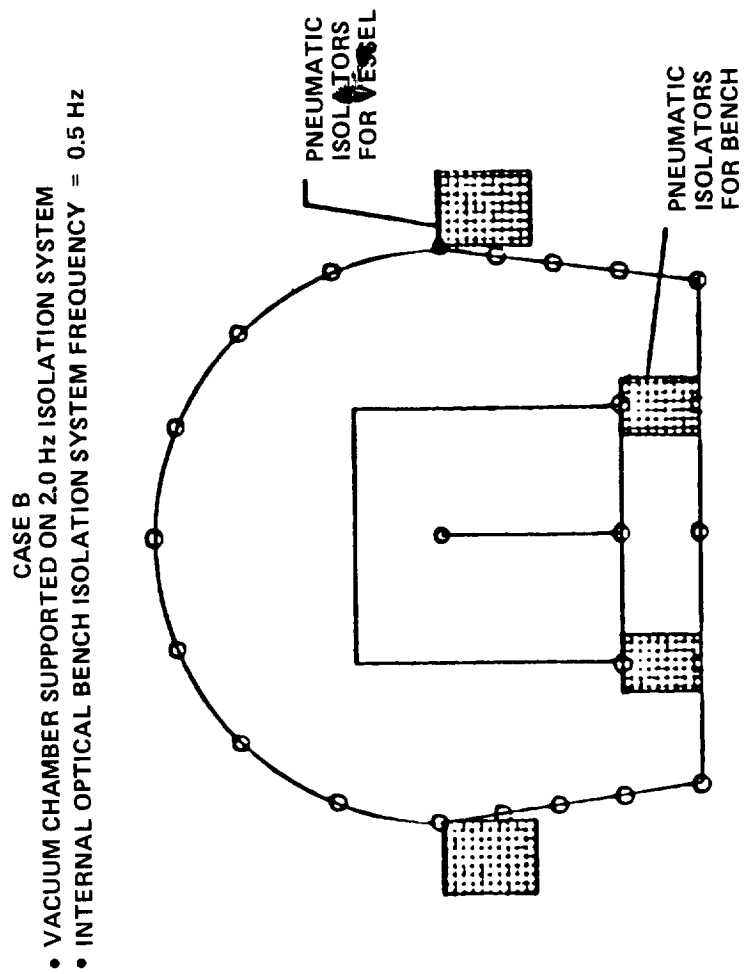


Figure 8. PSD 2-D Case B

- CASE C
- VACUUM CHAMBER SUPPORTED ON CONCRETE FOUNDATION
 - INTERNAL OPTICAL BENCH SYSTEM ISOLATION FREQUENCY = 0.5 Hz

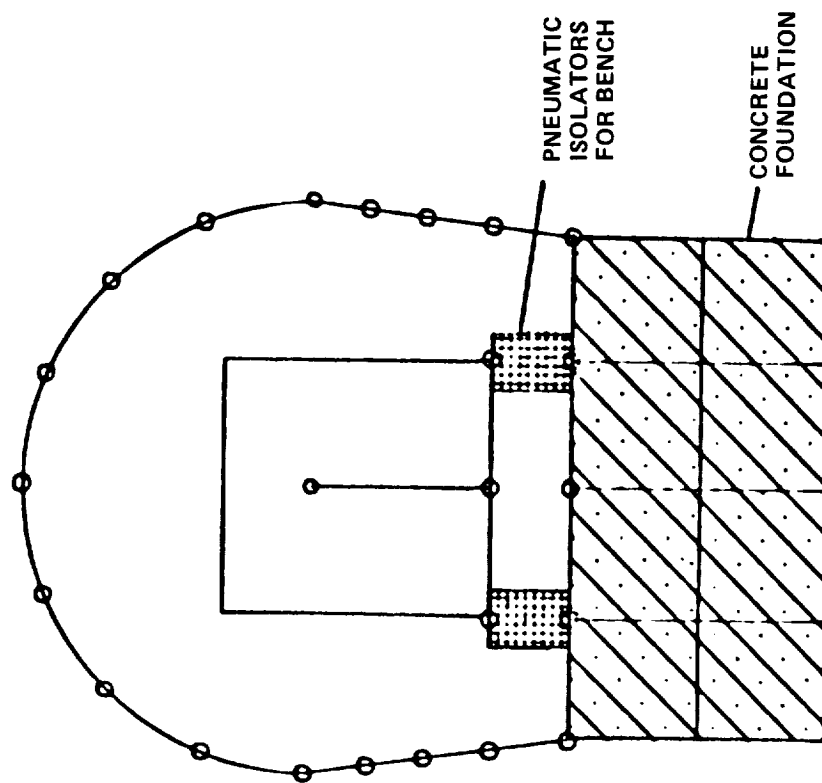


Figure 9. PSD 2-D Case C

- CASE D
- VACUUM CHAMBER SUPPORTED ON 1.0 Hz ISOLATION SYSTEM
 - INTERNAL OPTICAL BENCH ISOLATION SYSTEM FREQUENCY = 2.5 Hz

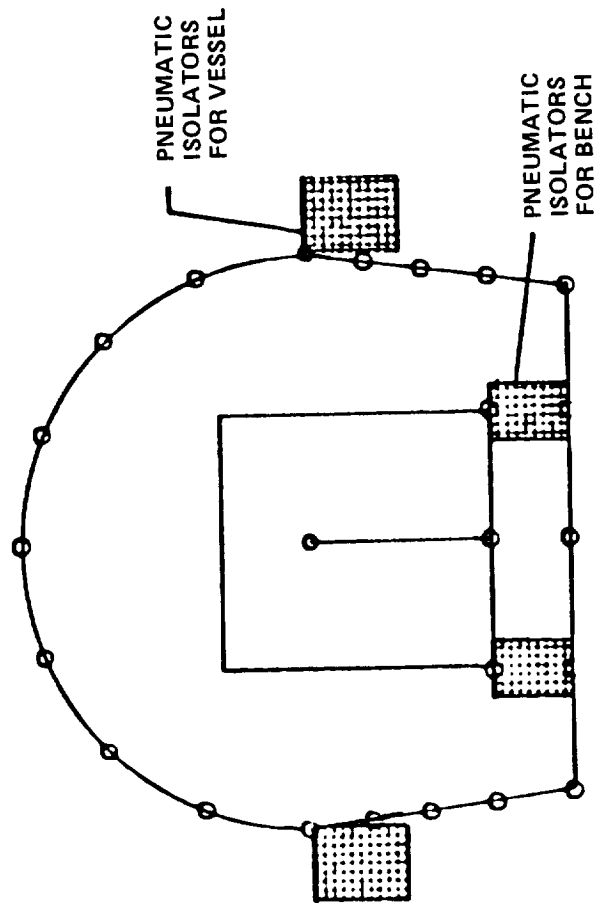


Figure 10. PSD 2-D Case D

- CASE E
- VACUUM CHAMBER SUPPORTED ON 1.0 Hz ISOLATION SYSTEM
 - INTERNAL OPTICAL BENCH ISOLATION SYSTEM FREQUENCY = 2.5 Hz

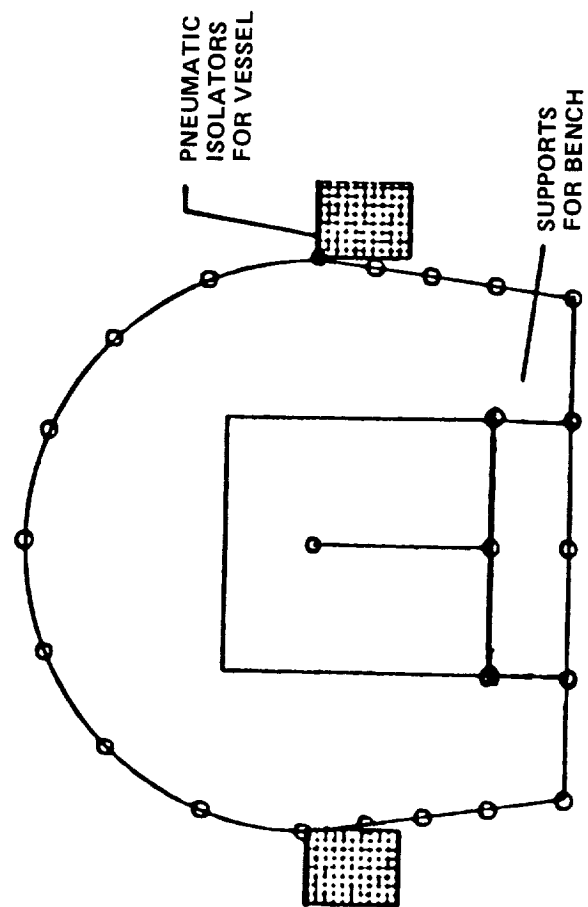


Figure 11. PSD 2-D Case E

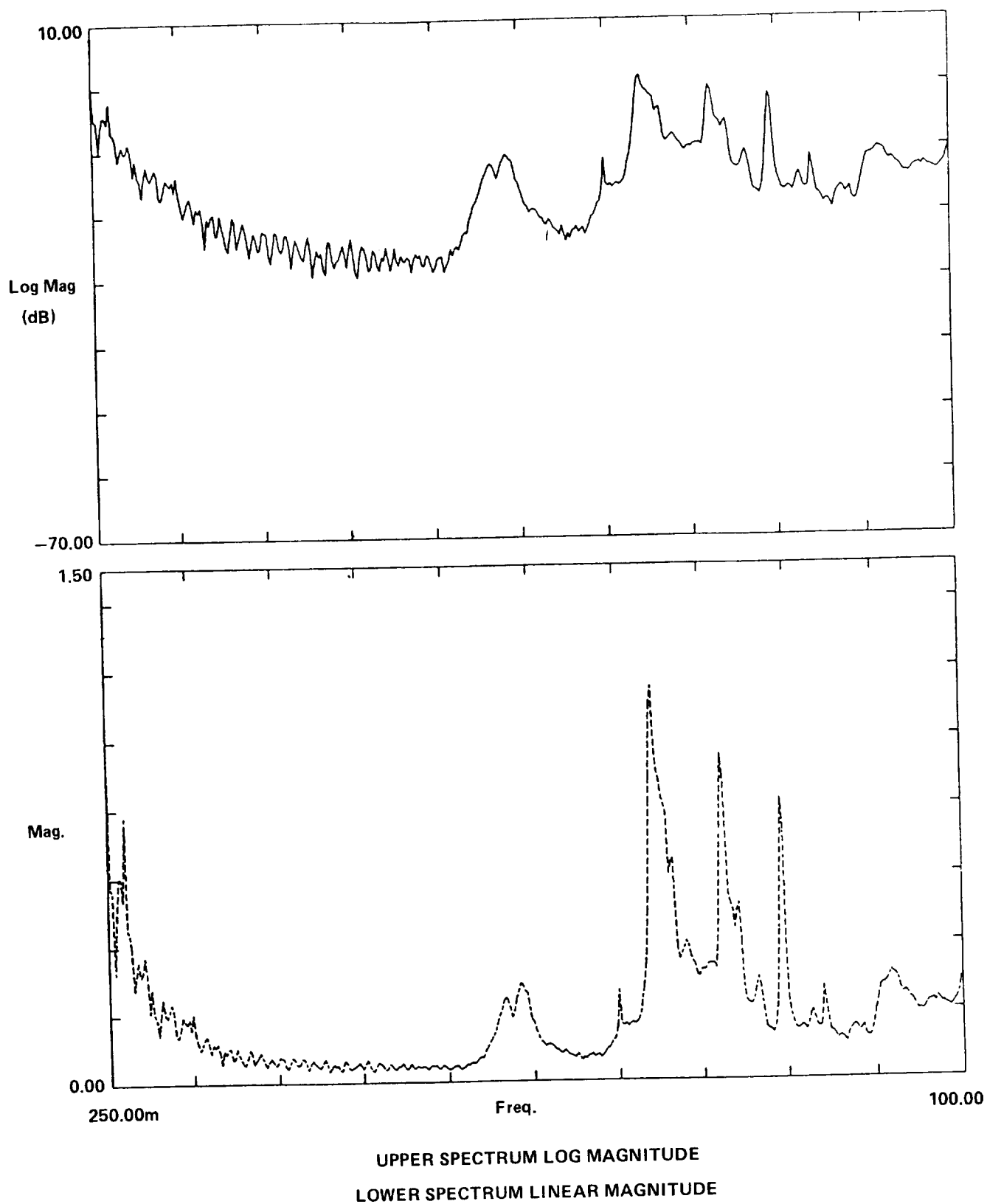


Figure 12. Model Peaks FRF Spectra, Shell and Base